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NUMERICAL STUDY CONCERNING THE INFLUENCE OF THE MATERIALS TO THE MESHING OF AN INVOLUTE HELICAL GEAR SET

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In gears transmissions, the load carrying capacity is a parameter with an influence for the design of the transmission. The main requirement is the steadily-increasing power density but this is due also to the materials used for the gear set. Gears are made of various types of materials, such as iron-based materials, nonferrous metals, or plastic materials, in accordance to their usage. The strength of gears differs depending on the type of material, heat treatment or quenching applied.

The purpose of the study in the following paper is to investigate the static behavior of the transmission systems with helical gears that are made from various materials. Analysis was carried out using a FEM software. Results show the gear`s behavior in the meshing contact. It also reveals the distribution of the pressure along the width of the teeth and the behavior due to bending stress.

Keywords: helical gears, gears material, static gears bending behavior, gears contact.

1. Introduction

Gears are mechanical elements used in the transmission of speed and power from one engine to another device. Nowadays, mechanical transmissions with gears are used for a wide range of applications (automotive industry, high power transmissions as turbines, pumps, windmills et alt.). In the past decades, engineers were focused on the transmission of power and torque [1].

In our days the main target for the designers is to increase the power, usually at a higher speed level and, at the same time, to reduce the size of mechanical transmissions. One of the solutions is directly linked with the choosing process of materials for the pinion and gear.

The problem is choosing optimum materials for gears exposed to bending in the region of the teeth root. Also, deflection and Hertzian contact in the

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meshing gears contribute to the difficulty in finding the proper materials required [2], [3]. Some researches consider that the primary indicator of working capacity is surface contact strength [4]. This is due to the behavior of the gears in dynamic conditions. One of the main sources of alteration is caused by surface contact damage or pitting. This last concept is defined by progressive chipping from small pits to wide holes with plastic deformation of the contact surfaces [5].

The present article treats a numerical study managed by the Finite Element Method. Thus, investigating the static behavior of the transmission systems with helical gears made from various materials. Analysis was carried out using a FEM software. Results show the behavior of gears set in the meshing contact, the distribution of pressure along the teeth width and what happens due to bending stress.

2. The model of the helical gears transmission

The importance of this study is finding the geometric model of the analyzed gears and also discovering the details of gear construction. The design model of the gears was created from the structure of the test rig. The ratio of the transmission is 3.06. The geometric model of the transmission is shown in Fig. 1.

The geometry of the cylindrical gear with helical teeth is based on the profile of the reference rack. [6].

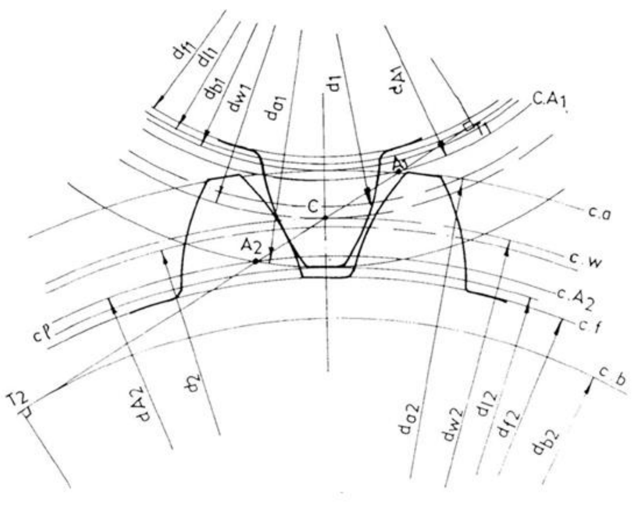

Fig. 1. Geometrical characteristics of the set of helical gears– adapted ISO 21771:2007, (en) [6]

Concerning the manufacturing of the gears, the Gear Generator and Form Cutting Method were opted. Form Cutting Method is a milling working method, with an inverse involute cutter profile. For milling process, with a disc miller a cylindrical tooth can be seen in Figure 2. For each case, the profile of these cutters changes depending on the module, gear angle and the number of teeth, and for gears with displaced profile teeth, depending on the displacement coefficient (figure 2).

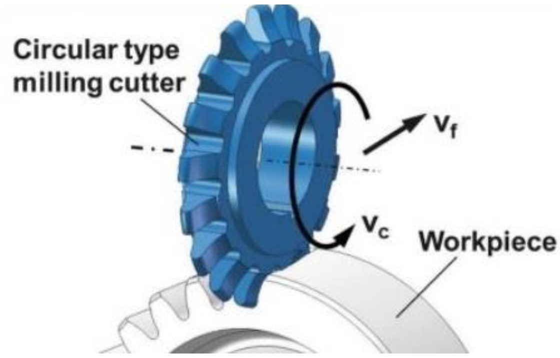

Fig. 2. Milling cutter for helical gears [7]

In table 1 are shown the main parameters of the geometry for the gear set.

Tabel 1

The main geometric parameters of the gear set [6]

ELEMENT	SYMBOL	U.M.	PINION	GEAR
MODULE (NORMAL)	m_n	mm	4	4
NUMBER OF TEETH	z	-	15	46
PRESSURE ANGLE	α	deg	20	20
HELIX ANGLE	β_d	deg	10	10
TRANSMISSION RATIO	U	-	$z_2/z_1 = 3.06$	
OUTSIDE DIAMETER	d_a	mm	72.269	193.662
BASE DIAMETER	d_b	mm	57.147	175.252
ROOT DIAMETER	d_f	mm	54.341	175.734
PITCH DIAMETER	D	mm	60.925	186.838
PROFILE SHIFT COEFFICIENT	x_t	mm	0.427	-0.138
GEAR FACE WIDTH	b	mm	70	20

The algorithm of the numerical modeling process includes the profile (designed with two profile curves, and the radius of curvature at the base of the tooth). Figure 3 presents the results of the model design.

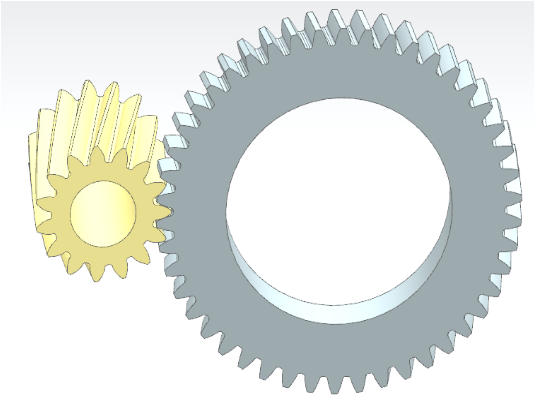

Fig. 3. The gear in mesh

The model was designed to simulate the static analysis under load of the gear as shown in Figure 4. To transfer the file from SIEMENS UNIGRAPHIX NX, from which the 3D model was created, in Ansys-Workbench, to perform FEM analysis, the model is saved in ACIS (SAT) format.

To simulate the gear contact, a 4-tooth sector of the respective pinion or gear is required. Once the file is imported into Ansys Design Modeler, it was applied to the general control. The model is shown in Figure 4.

The mesh of the automatic proposed finite element model is not sufficient for a balance between running time relative to simulation accuracy. The following meshing procedures will have 3 levels, on the knob, a general mesh, on the 3 faces of the tooth predominantly in contact, an intermediate mesh and on the contact area, an accurate division.

Therefore, it is obtained estimated values, both in the case of the deformation of the tooth on its entire length and in the fidelity of the contact pattern. The coordinates of the nodes are establishing for the profile of the teeth gears, depending on the number of nodes chosen in axial and radial direction on the rim, profile and radius on the teeth.

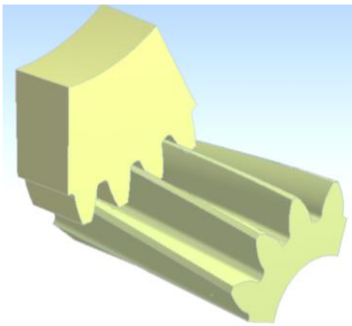

Fig. 4. The model of the meshing gears for the FEM analysis.

It is considered that the lateral surfaces of the bodies in contact keep the limit conditions, represented by the rigid surfaces, being located far enough from the contact. The nodes on the axis of the two components of the gear are considered the reference to control the movement.

3. Materials

We choose for our theoretical, numerical study, a set of three materials with known chemical composition, known mechanical properties and various ways of heat treatment.

Considering the heat treatment, several guidelines from literature [8], [9], includes recommendations for the case depth as a function of module. As mentioned, this study is about the case of high power transmission. In this case, the heat treatment, carburizing process, is relatively 1000 °C, affecting the gears to a nominal case depth range of 0.75 mm to a maximum 1.2 mm. This corresponds to a gear set to be manufactured with a module of 4 mm. It is followed by the compensation for grinding losses during hard machining of the carburized gear. It is supposed that MAT1 has a higher value of hardness at the contact (55-60 HRC) and the second material, MAT2, a lower value (50-55 HRC).

Finally, we chose three materials, MAT1, MAT2, MAT3 as a base for our study. The first two materials are used for carburized gears, materials with a near chemical composition but with a difference concerning mechanical properties (Tensile strength, R_{m}, and HRC hardness). The third material correspond to an aluminum alloy.

Tabel2

Chemical composition of materials [wt.%]

Material	C	Si	Mn	Cr	Ni	Mo	Vn
MAT 1	0.25	0.12	1.45	1.23	0.89	0.53	0.04
MAT 2	0.20	0.26	1.17	1.13	0.30	0.20	0.03

Tabel3

Mechanical properties of materials

PROPERTY	MAT1	MAT2	MAT3
YOUNG`S MODULUS [MPa]	2E11	1.93E11	7.1E10
POISSON`S RATIO	0.3	0.31	0.33
TENSILE STRENGHT [MPa]	2.5E8	2.07E8	2.8E8

4. Analysis

In the first step of this analysis, pitting strength and root bending stress will be calculated using international standards [10]. It will be calculated on the basis of the ISO 6336-2 [11] and ISO 6336-3 [12] standards.

The maximum contact stress:

$$\sigma_{HO} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b}\frac{u+1}{u}}, \quad (1)$$

$$\sigma_H = Z_B \cdot \sigma_{HO}\sqrt{K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha} \cdot} \le \sigma_{HP}, \quad (2)$$

Where σ_H is the calculated contact stress, σ_{HO} is the nominal contact stress, σ_{HP} is permissible contact stress. The terms are: Z_H–zone factor, Z_E – elasticity factor, Z_ε – contact ratio factor, Z_β – helix angle factor (pitting).

K_A - application factor, K_V - mesh load factor, $K_{H\beta}$- face load factor, $K_{H\alpha}$ - transverse load factor, Z_B - single pair tooth contact factors for the pinion.

F_t is the transverse tangential load at reference cylinder per mesh, d_1 is the diameter of the pinion, b is the face width, u is the gear ratio (z_2/z_1).

The bending contact strength:

$$\sigma_{FO-B} = \frac{F_t}{m_n \cdot b} Y_F \cdot Y_S \cdot Y_\beta, \quad (3)$$

$$\sigma_F = \sigma_{FO} \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \le \sigma_{FP}, \quad (4)$$

Where σ_F is the calculated bending stress, σ_{FO-B} is the nominal bending stress, σ_{FP} is permissible bending stress. The terms are: Y_F – tooth form factor, Y_S – stress correction factor, Y_β — helix angle factor (bending).

K_A - application factor, K_V - dynamic factor, $K_{F\beta}$ - face load factor (root stress), $K_{F\alpha}$ - transverse load factor (root stress)

F_t is the transverse tangential load at reference cylinder per mesh, m_n is the normal module.

All calculations are based on ISO 6336-2, ISO 6336-3 method of calculation, and they include the terms corresponding to the geometry for our case study and the materials chosen. The load is taken for the torque applied.

Concerning the numerical study, the behavior of the transmission under static load was simulated. Furthermore, the material properties chosen for each case of the system and the load were taken also into account. The load was established to a torque applied to the pinion head, in the head away from the gear.

The properties of the pinion and gear materials were considered to be the same on each simulation. Thus, the following sets of material were made: MAT1, MAT2, MAT3. The main properties are presented in the table 3. C3D8I three-dimensional solid elements and GAP elements are used [13]. The structure is composed of approximately 618611 nodes and 370782 elements, the size of the element on the tooth length being 1mm and in the contact area 0.2mm, figure 5.

The contact conditions were applied for the contact between two teeth. The contact is marked only on the active flank of the gear. The menu for the contact settings in the presented model is entered from the ANSYS menu and it allows the choice of the contact type between surfaces. For this simulation we chose to use contact with friction, the coefficient of friction being 0.3.

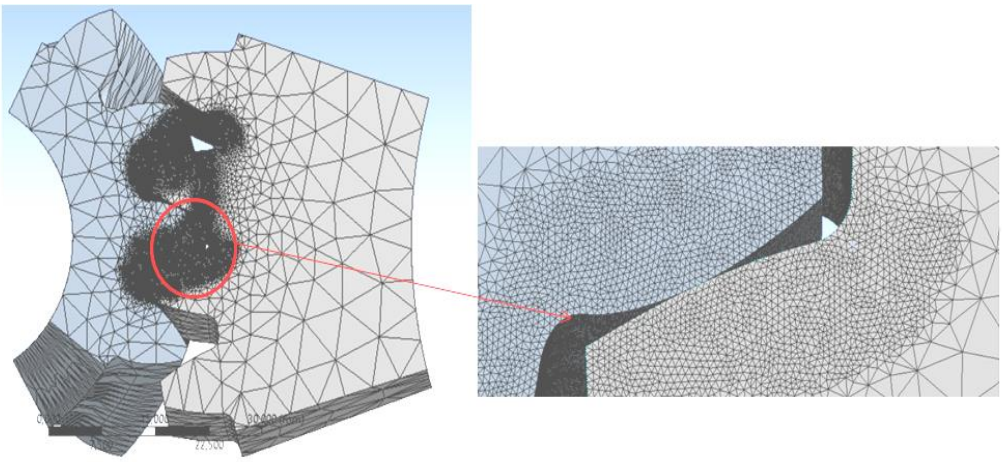

Fig. 5. FEM discrete model in mesh

5. Results

In the analysis we observed the following aspects: due to the deformations in the elastic domain of the system, we can see in the analysis the effect of bending teeth in contact, the effect of Hertzian contact (local) between of the two pairs of teeth in contact and structural displacements.

The displacement will be taken into account for the gear sectors, shown in figure 4. The displacement of the entire structure is due both to the moment applied on the pinion head and to the displacement imposed on the gear to simulate several gear moments for the evolution of the contact pattern.

Although it is a static analysis, we consider performing the analysis in real operating conditions. For this we initially study the total deformation of the structure, as shown in Figure 6.

Based on the different materials, a variation of the range of values along the length of the tooth in contact is observed. In the following table it can be seen the values of the deformation of the tooth in contact.

For an easier analysis of the results, the following graph was introduced. In it, the x-axis represents the sectors along the tooth where the measurement was evaluated.

The deformation values were measured at a height of 33.88 mm from the axis of rotation of the part, in radial direction, from the hub.

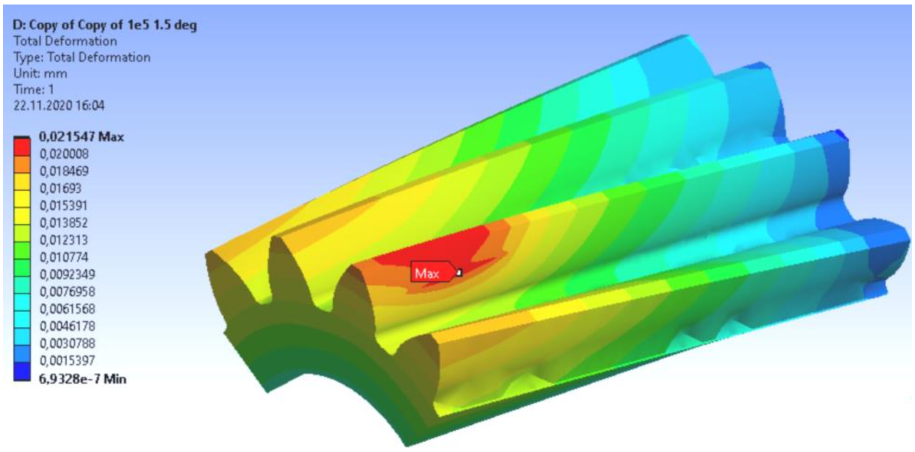

Fig. 6. Total deformation

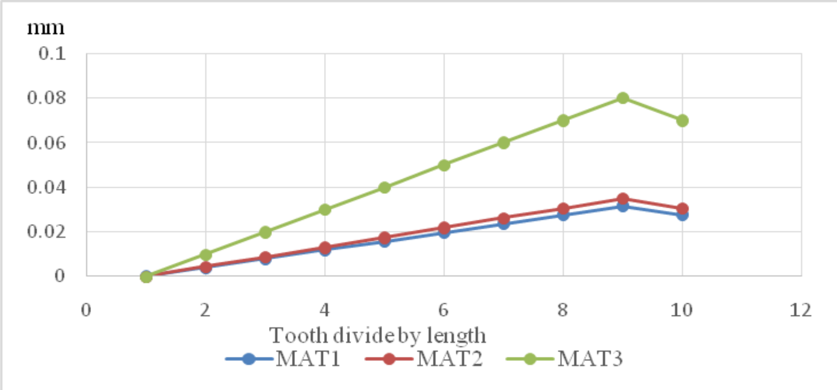

Fig. 7. Deformation along the tooth

The behavior of the teeth under load is analyzed both in the contact area and at the tooth root. We present the stress states in both areas for each material. This state of stress shows variations depending on the material and is closely related to the gear meshing position.

As it can be seen, in the Figures 7, 9 and 10 for MAT1 and MAT 2, that offer the best performance under the conditions imposed at a load of torque T=200 Nm, the stress is approximately 1154 MPa (σ_H), in the area of contact between the teeth and 140 MPa at the tooth root stress (σ_F). The reference for the validation of the study is the analytical calculation results. The result for analytical calculation for maximum contact stress, conform eq. (1), (2). is σ_H = 1232 MPa The result for analytical calculation for maximum root stress conform eq. (3), (4) is σ_F = 187 MPa.

The influence of materials on the Von Misses stress field along the root of the teeth is presented in the following table and is better taken into account by numerical study. As it can be seen, the softer the material, the lower the stress in favor of deformation.

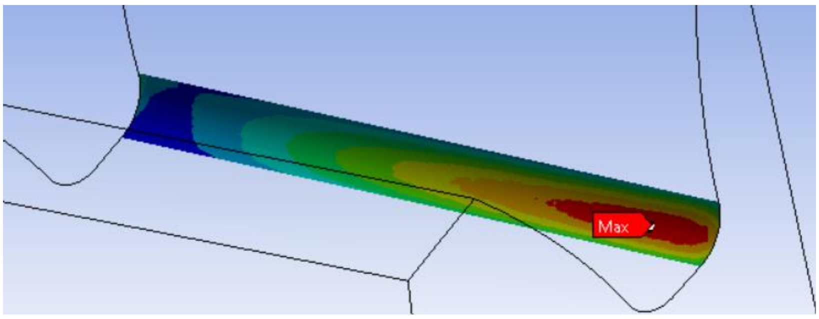

Fig. 8. Tooth root stress

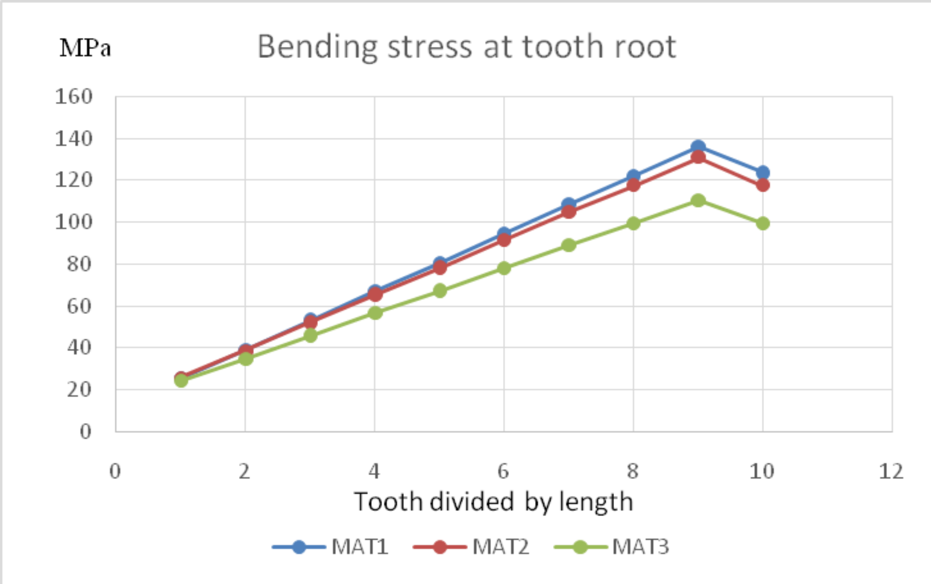

Fig. 9. Equivalent Stress root teeth gear

Figure 10 shows the Equivalent Stress at that load level T=200 Nm. The value of the 1154 MPa as a local contact stress is close to a real one in the same loading conditions for the same material, and the same geometry.

That means the result is better than the result obtained with analytical calculus due to the 3D model in mesh, with the consideration of the real contact along the teeth (one or two teeth in contact) and the real mesh along the helix orientation of the teeth contact.

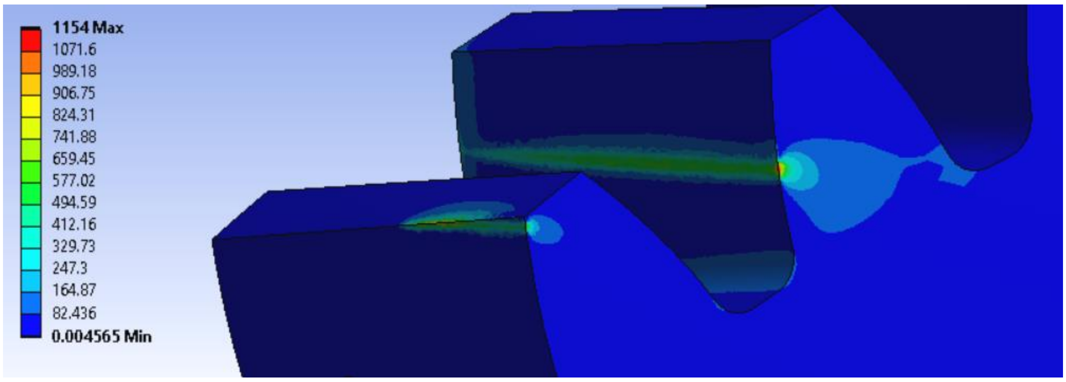

Fig. 10. Hertz contact stress analysis MAT 1

Taking into account the contact within the gear, we can analyze the pressure distribution, related to the gear moments. Thus, a maximum of the contact pattern on MAT2 is observed, which offers the best behavior, of 2 microns, similar to that of MAT1, being close materials.

On the other hand, the MAT3 model suffers a deformation of almost 6 microns, thus leading in time to the destruction of the piece.

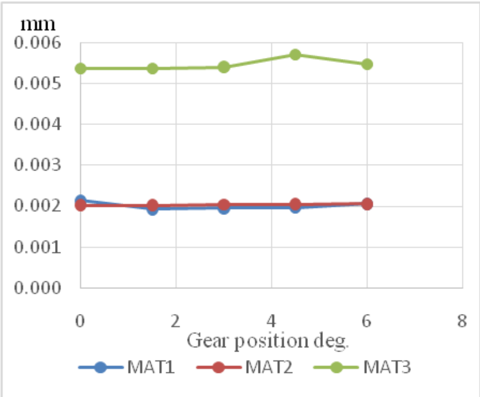

Fig. 11. Penetration

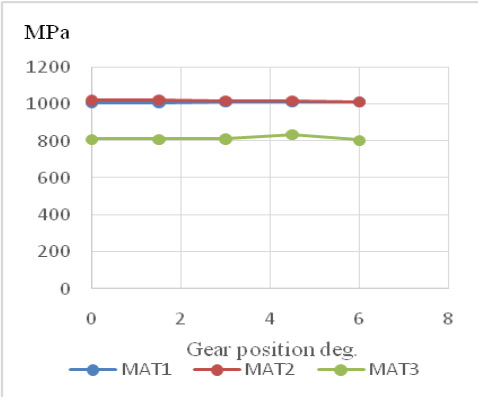

Fig. 12. Contact pressure variation at different angular positions of the gear.

6. Conclusions

- In this paper we designed two separated helical gears (pinion and gear) and a set of a gear mechanical transmission.
- Gear-bearing materials need a balance between Young's modulus and tensile strength. If only the Young module is high, we have small deformations but significant stresses. The first two materials used for the study are almost the same chemical composition but a different heat treatment.
- Of the three materials chosen, an optimal behavior is presented by MAT2, balancing the two required demands, the tooth deformation and the tooth mechanical properties, contact stress and root stress.
- It can be observed, MAT1, MAT2 have a good behavior at the contact pressure in the correlation with the Hertzian deformation (small deformation).
- The third material, MAT3, used only for an evaluation of the stress and deformation domain, has a capacity for large deformation, but lower state of stress at the root region. Of course, due this large deformation this type of material will not be recommended for important loads (torque).

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